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RESEARCH MEMORANDUM

EXPERIMENTAL INVESTIGATION OF THE VIBRATION CHARACTERISTICS

OF FOUR DESIGNS OF TURBINE BLADES AND OF THE EFFECT

PRODUCED BY VARYING THE AXIAL SPACING BETWEEN

NOZZLE BLADES AND TURBINE BLADES

By W. C. Morgan and C. R. Morse

Lewis Flight Propulsion Laboratory
~~CLASSIFIED~~ ~~CLEVELAND, OHIO~~ ~~CHANGED~~To: ~~Confidential~~~~Re: #1646~~By authority of ~~John W. Crawley~~ Date ~~7-21-53~~

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SUMMARY

An experimental investigation was made to determine the effect of varying the spacing between the nozzle blades and the turbine blades on turbine-blade vibration for four turbine-blade designs of different degrees of stiffness. The turbojet engine used in the investigation was operated at three conditions of nozzle-blade-to-turbine-blade spacing.

In general, there was a tendency toward increase in occurrence of vibration with decrease in spacing. The effect was most evident in the case of the turbine blades that had greater stiffness. Vibration in the lower-frequency modes was almost negligible for these blades at the condition of maximum spacing; when the spacing was decreased, the occurrence of lower-frequency mode vibrations increased to a considerable degree.

INTRODUCTION

Failure of turbine blades imposes serious limitation on the service life of turbojet engines, and it is of prime importance that the causes of turbine-blade failure be determined. Previous research into the mechanism of turbine-blade failure has indicated that vibration may be one of the more significant factors that affect the service life of turbine blades (references 1 to 4). During the course of a recent investigation (reference 4) a number of turbine-blade fractures displayed definite visual and metallurgical evidence of fatigue failure attributable to vibration.

The experimental investigation described in this report was conducted to determine the effect on turbine-blade vibration produced by changing the relative axial spacing between the nozzle blades and the turbine blades and to evaluate the effect of different degrees of

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turbine-blade stiffness. Turbine-blade vibration was observed during engine operation for three axial positions of the turbine blades relative to the blades of the nozzle diaphragm. The vibration characteristics of four turbine-blade designs were determined under each condition of spacing. The individual turbine-blade designs were essentially similar in respect to performance considerations, the principal difference among these several designs being in the degree of inherent cantilever stiffness.

The vibration characteristics of the turbine blades were determined through the use of high-temperature strain gages mounted on the turbine blades. The data obtained were used to determine the frequency of vibration and the stress in the location of the strain gages. These data were collected for the turbine-speed range from idling to the rated maximum. The scope of this report was confined to the determination of blade vibration; no performance data were taken.

EXPERIMENTAL EQUIPMENT AND PROCEDURE

It was necessary to modify the turbine section of the engine in order to change the spacing between the nozzle blades and the turbine blades.

The high-temperature strain gage and the instrumentation required for observation and recording of strain-gage signals were of the type described in references 5 and 6. Some recent improvements on the high-temperature strain gage and modifications of the turbine are discussed in the following paragraphs.

Description of engine. - A representation of the turbojet engine used in the investigation is shown in figure 1. The turbine section consisted of a single-stage rotor with eight cylindrical combustors mounted symmetrically around the aft frame; there were 64 equally spaced stator blades in the nozzle assembly.

Modification to engine. - The sketch shown in figure 2 indicates the modification to engine components made in order to provide a passage for lead wires from the strain-gage terminal connection on the aft face of the turbine wheel to the slip-ring rotor at the forward end of the engine. Axial holes were bored through the turbine shaft bolt, the compressor shaft, and the special slip-ring drive shaft in the accessory section.

The three variations in axial spacing between nozzle blades and turbine blades were designated as position 1 ($1\frac{1}{4}$ in.), position 2 (1 in.), and position 3 ($\frac{3}{4}$ in.). These distances were measured from the trailing edge of the nozzle blades to the forward faces of the turbine-blade bases. The changes in spacing were obtained for positions 1 and 2 by

relocation of the turbine rotor in an axial direction. The nozzle assembly was moved aft in order to decrease the spacing to the requirement for position 3.

Description of turbine blades. - The physical characteristics of the four designs of turbine blades are presented in table I. The designs were designated A, B, C, and D. Designs A and B were essentially similar, with the exception that in design B the trailing edge had more taper. Design C was more massive than any of the other designs. The design D blade represents a refinement of the previous three designs. Designs C and D were inherently stiffer than designs A and B.

Strain-gage instrumentation. - The data were obtained from high-temperature resistance-wire strain gages mounted on the turbine blades. The techniques of fabrication and mounting of these strain gages are described in references 5 and 6. A sketch of a strain gage prior to mounting is shown in figure 3.

Some advances that have resulted from recent research in the field of high-temperature strain gages are described in reference 7. Among these improvements are a more reliable cement for bonding the strain-sensitive wire to a turbine-blade surface, a more reliable electric connection between the filament and the lead wires, and a new type of wire for the strain-sensitive filament. In the investigation reported herein, the turbine-blade vibration data were obtained from strain gages composed of the following materials: National Bureau of Standards L-6AC precoat; Quigley AAA No. 1925 cover ceramic; 0.001-inch diameter Karma filament wire; and 0.010-inch-diameter 80-percent platinum - 20-percent iridium tubular lead wires.

A photograph of an instrumented turbine blade is shown in figure 4. The strain gage was located near the base on the concave side at the trailing edge. After insertion of the blade in the turbine rotor, the lead-wire conduit was secured to the aft face of the rotor by means of spot-welded straps. Two similar straps were welded to the blade base and the rotor rim to ensure a positive electric connection at this point of the strain-gage circuit.

Three turbine blades of each design were used in the investigation. A complete installation is shown in figure 5. The results from only one turbine blade of each design are presented in this report; the additional instrumented blades were provided in order to check the validity of strain signals.

The lead wires were secured at the terminal plate mounted on the aft face of the turbine rotor. These terminals were connected to the slip-ring unit shown in figure 6.

Experimental procedure. - The procedure consisted in operating the engine in a sea-level test stand through a speed range from 2000 (idling) to 8100 rpm (full speed). As speed was increased, strain-gage signal outputs were held under observation on oscilloscope screens. With the appearance of a signal that indicated vibration, turbine speed was adjusted until the oscilloscope trace showed maximum amplitude of vibration. All signals then were recorded on an oscillograph. These records were used in computation of frequency and stress (details of the methods are given in reference 5).

The foregoing procedure was followed for each installation of the four turbine-blade designs examined in the course of the investigation. Vibration characteristics were determined for each turbine-blade design under the previously described conditions of relative spacing.

RESULTS AND DISCUSSION

The results of the investigation are presented in figures 7 to 10. These critical speed diagrams show the data obtained for each of the four designs of turbine blades at the three axial locations relative to the nozzle blades. The ordinate scale is the frequency of vibration; the abscissa represents turbine speed. Occurrence of vibration and magnitude of stress range (double amplitude) in the location of the strain gages are indicated by appropriate symbols. Order lines represent the loci of points where the frequency is a definite multiple of the turbine speed. These reported data were confirmed by the data obtained from the check blades.

The order lines shown, 8, 16, 64, and 128, are those predictable from an analysis of the geometry of the engine. The eighth-order excitation would be produced by the combustion chambers; the sixteenth-order excitation, by the second harmonic of this source. Similarly, the sixty-fourth order excitation would be ascribable to the nozzle blades; the second harmonic of this source is represented by order line 128.

Design A

The critical speed diagrams for positions 1, 2, and 3 of turbine blade A (fig. 7(a), 7(b), and 7(c), respectively) show that there was an appreciable effect on vibration as a result of the change in spacing. In position 1, the first-bending mode was the predominant vibration. Only one case was observed of excitation of the first-torsional mode. There were several occurrences of complex-mode vibration.

The first-bending mode was susceptible to excitation at many orders of turbine speed not predictable from an analysis of engine geometry. The first-torsional mode appeared near the second harmonic of the

combustors. The fundamental order of the nozzle blades excited the more complex modes.

In position 2, a comparatively lesser number of vibrations were observed for the first-bending mode. There was an increased tendency toward vibration in the first-torsional mode, but the complex-mode vibrations were not affected to any considerable extent with the exception that one occurrence was observed of vibration at the second harmonic of the nozzle blades. Vibration in the first-bending mode appeared at close increments of speed when the turbine blade was operated in position 3. There was an increase in the occurrence of complex-mode vibration attributable to the fundamental order of the nozzle blades. Vibration was observed, also, at the second harmonic of the nozzle blades. There was an absence of detectable vibration in the first-torsional mode.

For the first-bending mode, a trend toward increase in stress level was observed with decrease in nozzle-blade-to-turbine-blade spacing. The peak stress for first-torsional mode vibration occurred at position 2. This was true, also, for the complex-mode vibration. These results must be considered only qualitative, however, because of several factors that will be discussed in a subsequent part of this report entitled FACTORS AFFECTING INTERPRETATION OF STRAIN-GAGE DATA.

Design B

The critical speed diagrams obtained for this turbine blade at positions 1, 2, and 3 (fig. 8(a), 8(b), and 8(c), respectively) indicate that the vibration characteristics of the design B blade were generally similar to those described in the case of turbine blade A. In position 1, the first-bending mode was excited into vibration at orders of turbine speed not attributable directly to engine configuration. Vibration in the first-torsional mode occurred at only one speed, probably as a result of second-harmonic combustor excitation.

When the turbine blade was operated in position 2, the vibration characteristics were not altered to any considerable extent with the exception that two cases of complex-mode vibration were excited by the second harmonic of the nozzle blades. It will be noted that the design A turbine blade displayed a characteristic nearly similar to that of blade B after the initial spacing had been reduced to that of position 2.

The design B turbine blade displayed less change in vibration characteristics with decrease in spacing than any of the other three designs. The occurrence of vibration in position 3 was not markedly different from that observed in the case of position 2. High-frequency vibration occurred only once at the second harmonic of the nozzle blades.

For vibration in the first-bending mode, the stress level increased with the change from position 1 to position 2. There was no significant difference in the stress levels measured at position 2 and position 3. The peak stress for first-torsional mode vibration occurred in position 2. In the case of complex-mode vibration, the peak stress occurred at the position of maximum spacing.

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Design C

The critical speed diagrams for the design C turbine blade in positions 1, 2, and 3 are shown in figures 9(a), 9(b), and 9(c), respectively. These figures indicate that the progressive decrease in spacing had a considerable effect on the vibration characteristics. In position 1, vibration was almost absent in the first-bending mode. Only one case was noted of this mode of vibration at the first order of excitation attributable to the combustors. There were two cases of vibration in the first-torsional mode. The remainder of the vibrations were complex modes ascribable to the fundamental nozzle-blade order of excitation, with one exception of high-frequency vibration caused by second-harmonic nozzle-blade excitation.

When the turbine blade was operated at position 2, the first-bending mode was excited into vibration by orders ranging from 8 to 16, inclusive. This increase of first-bending mode vibration was the principal change effected by the decrease in spacing from position 1 to position 2. The response to excitation of the first-torsional and more complex modes of vibration was not changed to any considerable extent by the decrease in spacing.

Operation of the turbine blade in position 3 was observed to have changed the vibration characteristics in the first-torsional mode of vibration. Several occurrences of this mode were not directly attributable to engine configuration. In addition to this change, one case was observed of vibration in a mode of a frequency slightly higher than the first-torsional mode. This mode of vibration did not occur during previous operation of the design A and design B blades or of the design C blade in position 1 and position 2. The complex-mode vibrations observed were similar in nature to those excited at positions 1 and 2 by the fundamental and second-harmonic orders of the nozzle blades.

In the first-bending mode, the highest stress occurred at minimum spacing; in first-torsional mode, at maximum spacing; and in the complex modes, also at the condition of maximum spacing.

In regard to the general effect produced in stress level by decrease in spacing, the trend was toward lower stresses rather than toward higher, as was observed for designs A and B. This effect was most marked in the initial change from position 1 to position 2.

Design D

The progressive decrease in spacing was accompanied by changes in vibration characteristics that were similar, in general, to those observed in the case of the design C turbine blade. The critical speed diagrams for positions 1, 2, and 3 are shown in figures 10(a), 10(b), and 10(c), respectively.

In position 1, the first-bending mode of vibration was excited at only one speed and by the second-harmonic order of the combustors. The first-torsional mode was not observed to occur. These were the principal disparities between the vibration characteristics of the designs C and D blades. Complex modes appeared as a result of excitation by the fundamental and second-harmonic orders of the nozzle blades.

When the turbine blade was operated in position 2, a considerable change was produced in the vibration characteristics. Numerous occurrences of first-bending mode vibration were observed, and one instance occurred of vibration in the first-torsional mode.

In addition, a complex mode of a frequency slightly higher than that of first-torsional mode vibration occurred at three turbine speeds. Higher-frequency complex modes were excited by the fundamental and second-harmonic orders of the nozzle blades.

The change produced by decreasing the spacing to position 3 was, to some extent, parallel to that observed for the design C turbine blade. It was characterized principally by the increase in susceptibility of the turbine blade to excitation in the first-torsional mode at orders of turbine speed not directly ascribable to engine geometry. The complex modes appeared as results of fundamental and second-harmonic nozzle-blade excitation. The first-bending mode of vibration was observed to occur at many orders of turbine speed.

For the first-bending mode, stress was highest at the condition of intermediate spacing; for the first-torsional mode, the peak stress occurred at minimum spacing. For the complex modes, peak stress occurred at maximum spacing.

The general level of stress appeared to decrease as the spacing was made progressively less between the nozzle blades and the turbine blade. The effect was not so marked as in the case of the design C turbine blade.

FACTORS AFFECTING INTERPRETATION OF STRAIN-GAGE DATA

There are several factors that merit consideration in an evaluation of the results. Among the more important of these factors is the significance that may be attached to the stress values reported.

Previous laboratory research had established the fact that the strain-gage location selected for the purposes of the reported investigation was optimum for the detection of vibration in the first-bending and first-torsional modes. In the case of detection of vibration in the complex modes, the possibility exists that the strain gage was not in an optimum location.

Excitation of vibration at orders of turbine speed not ascribable to engine configuration may be attributed to variations in mass flow and differences in velocity profile around the exit annulus of the nozzle diaphragm. This phenomenon has been observed in previous investigations (references 5 to 7). When conditions of operation were favorable toward first-bending mode vibration at a large number of orders of excitation, it was observed that there was present at all times a tendency toward low-amplitude vibration in this mode. In many cases, this type of vibration was observed to occur simultaneously with higher-frequency complex-mode vibration.

Another factor that must be given attention in the consideration of the results is the occurrence of higher-frequency vibration at close increments of turbine speed, specifically, those vibrations attributable to fundamental nozzle-blade excitation. The large number of higher-frequency complex-mode vibrations were not necessarily attributable to natural resonance; it is more probable that these vibrations were induced by vibration in adjacent turbine blades that had somewhat different natural frequencies.

SUMMARY OF RESULTS

An experimental investigation was made to determine the effect of varying the spacing between the nozzle blades and the turbine blades on turbine-blade vibration for four turbine-blade designs of different degrees of stiffness. In the following detailed summary, it was found convenient to consider the four turbine-blade designs in two groups, on the basis of nearly similar characteristics.

Designs A and B

1. First-bending mode vibration occurred at many orders of turbine speed for all three positions of axial spacing between nozzle blades and turbine blades. The peak stress increased as the spacing was decreased.

2. Vibration in the first-torsional mode was excited at the maximum and intermediate conditions of spacing, with the higher stress level occurring at the intermediate location. Further decrease in spacing resulted in the disappearance of this mode of vibration for the design A blade.

3. Complex-mode vibration excited by the fundamental order of the nozzle blades occurred at many different turbine speeds under all conditions of spacing. The peak stress occurred at the intermediate position for the design A blade and at the position of maximum spacing for the design B blade.

4. High-frequency vibration attributable to the second order of nozzle-blade excitation occurred only in the intermediate and minimum conditions of spacing. The peak stresses were higher at the intermediate position.

Designs C and D

1. There was only one occurrence of first-bending mode vibration for each design of turbine blade in the position of maximum spacing. Decrease in spacing introduced a large number of vibrations in first-bending mode. The highest stress occurred at minimum spacing for blade design C and at intermediate spacing for blade design D.

2. The response to excitation in first-torsional mode vibration increased with decrease in spacing. There was no occurrence of this mode at maximum spacing for the design D blade. The highest stress occurred at the maximum spacing for the design C blade and at the minimum spacing for the design D blade.

3. The occurrence of complex-mode vibration was not affected to any considerable extent by the progressive decrease in spacing. Vibration in this mode occurred at many turbine speeds. The highest stresses for both turbine blades were observed under the condition of maximum spacing.

4. The second order of nozzle-blade excitation caused complex-mode vibration of high frequency at every position. The stress levels decreased with decrease in axial spacing.

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TABLE I - PHYSICAL CHARACTERISTICS OF FOUR DESIGNS
OF TURBINE BLADES

	Design			
	A	B	C	D
Weight (grams)	230.3	226.4	292.5	275.3
Chord tip (in.)	1.720	1.636	1.763	1.662
Chord base (in.)	1.767	1.788	1.925	1.900
Maximum thickness at tip (in.)	.113	.116	.213	.081
Maximum thickness at base (in.)	.321	.321	.340	.347
Minimum thickness at tip (in.)	.024	.040	.058	.052
Minimum thickness at base (in.)	.040	.050	.085	.095
Total length (in.)	4.418	4.360	4.379	4.342
Weight of blade less root (grams)	140.3	136.4	202.5	185.3

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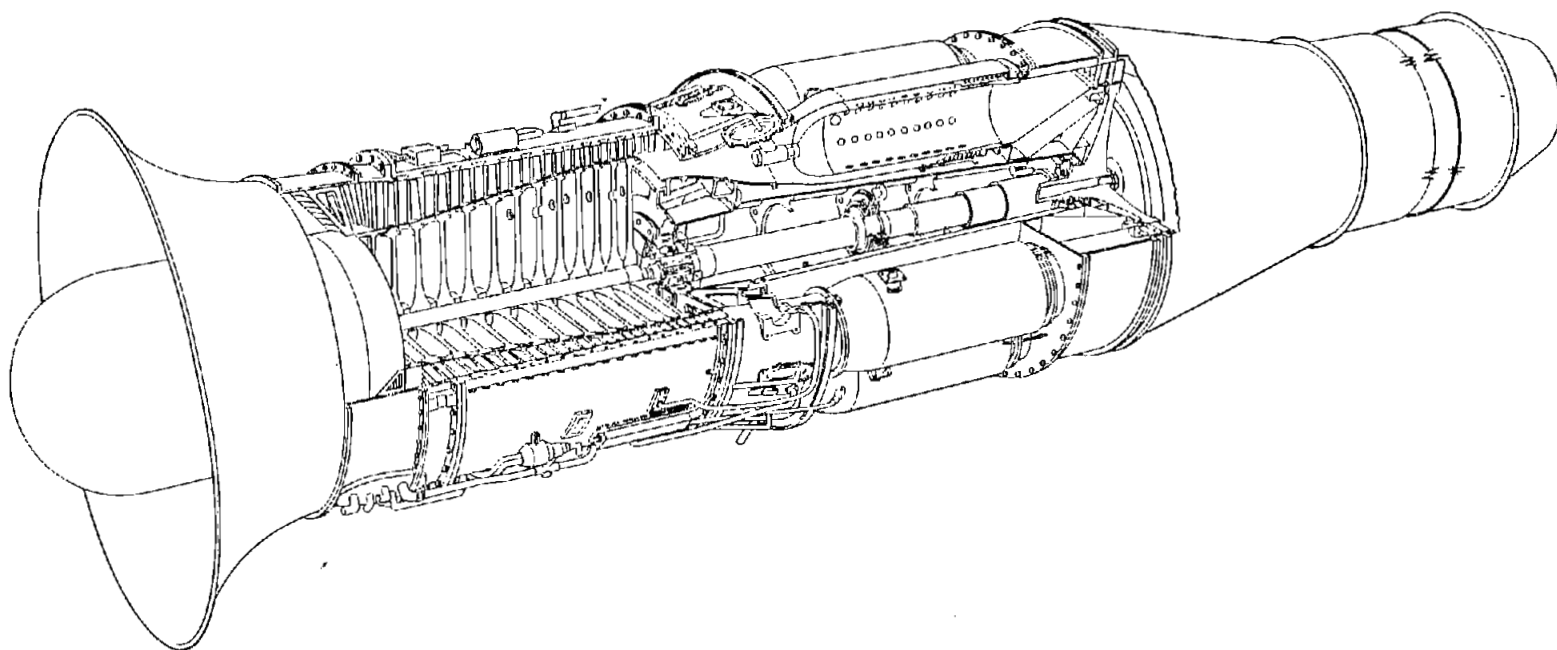


Figure 1. - Cutaway sketch showing principal components of turbojet engine.

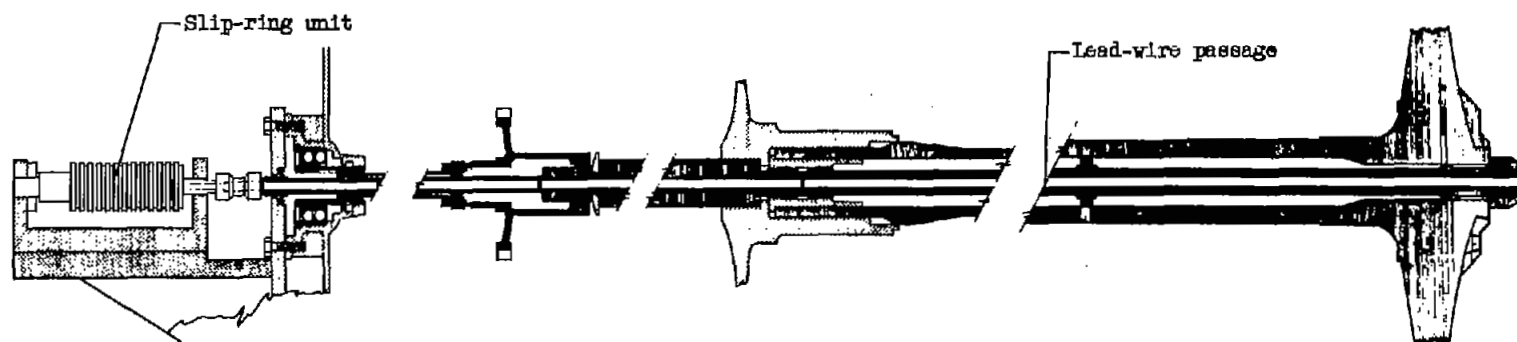


Figure 2. - Sketch of alteration to engine components for lead-wire passage and slip-ring drive shaft.

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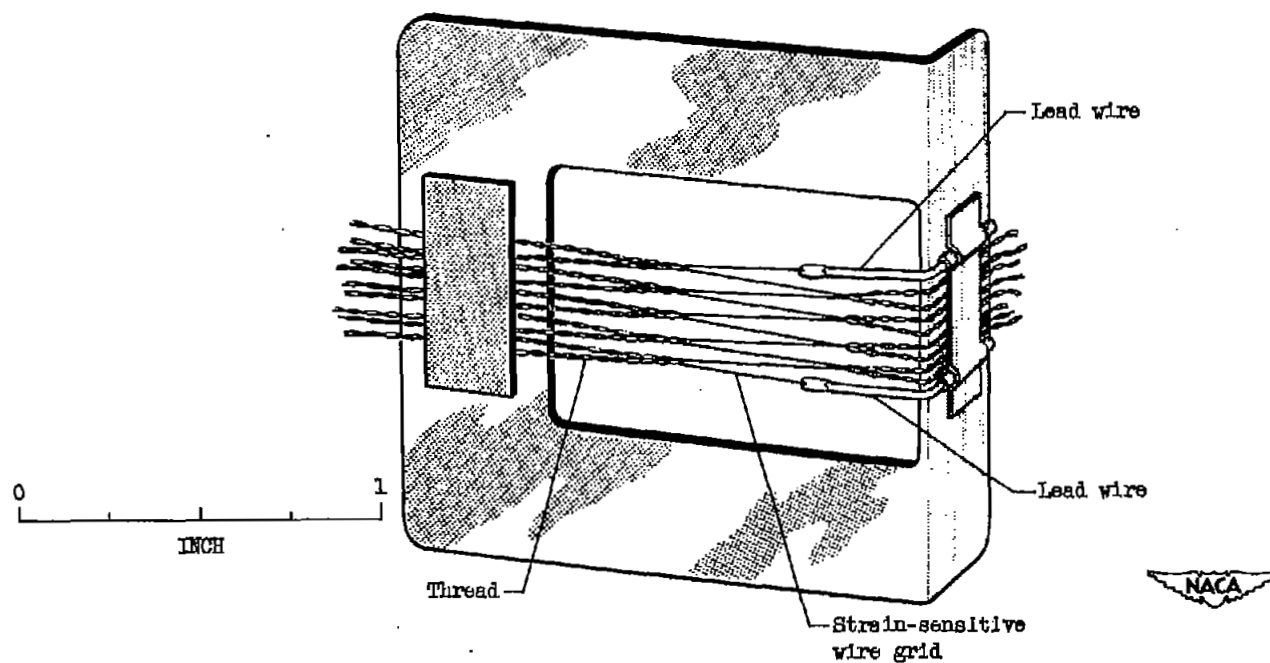


Figure 3. - High-temperature wire-resistance strain gage in frame ready for mounting.

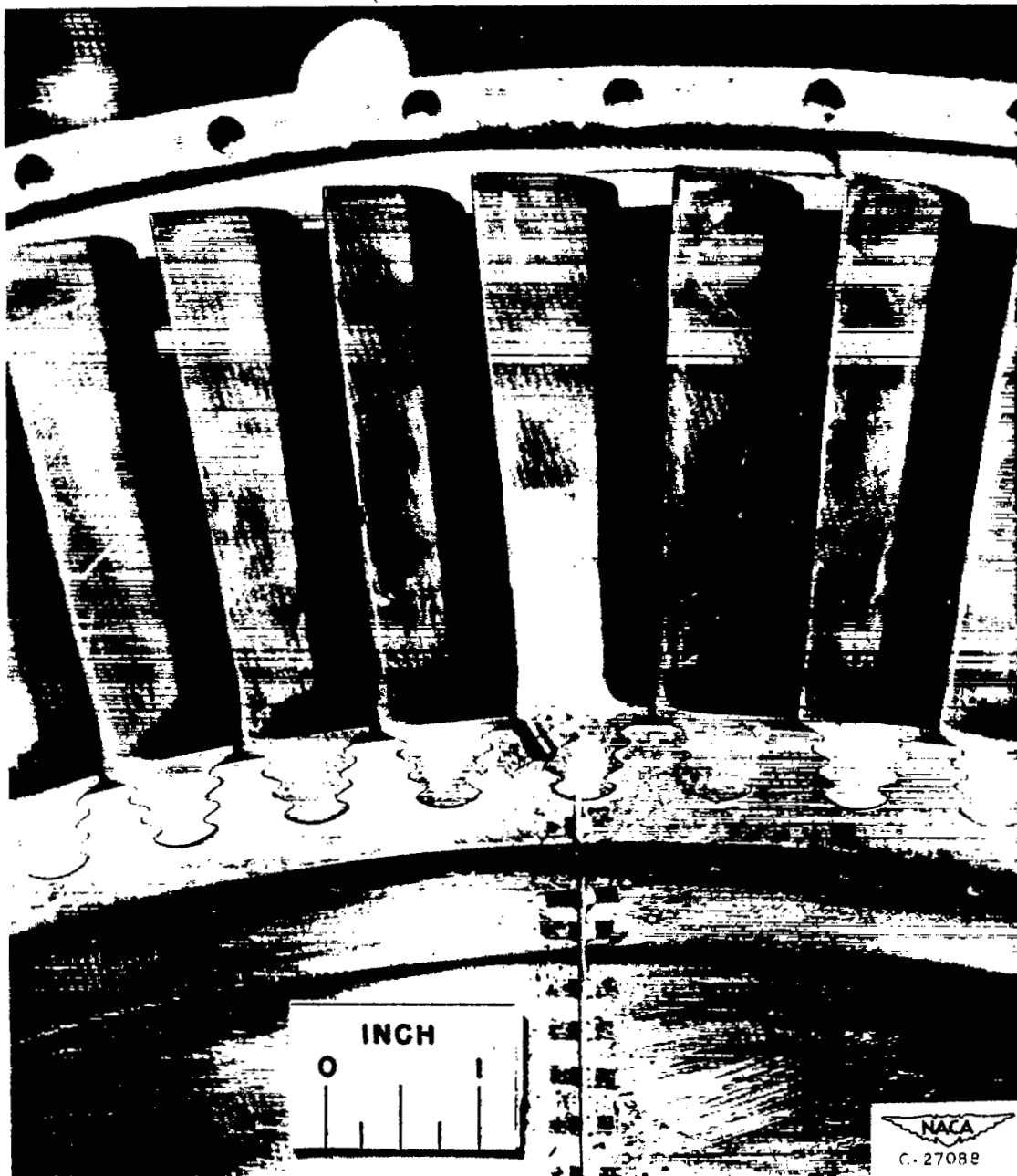


Figure 4. - Instrumented turbine blade installed in turbine wheel.

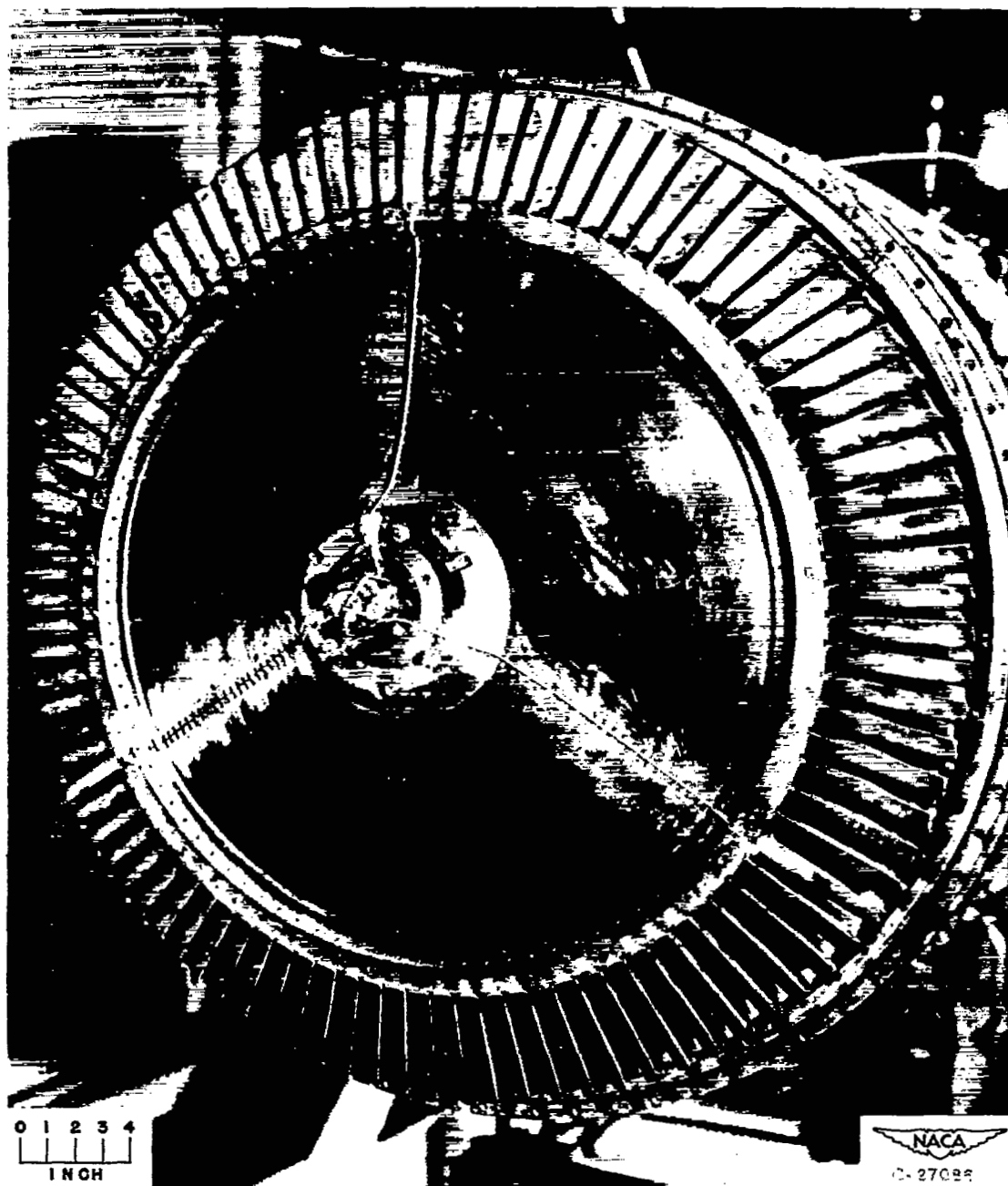


Figure 5. - Complete installation of three instrumented blades.

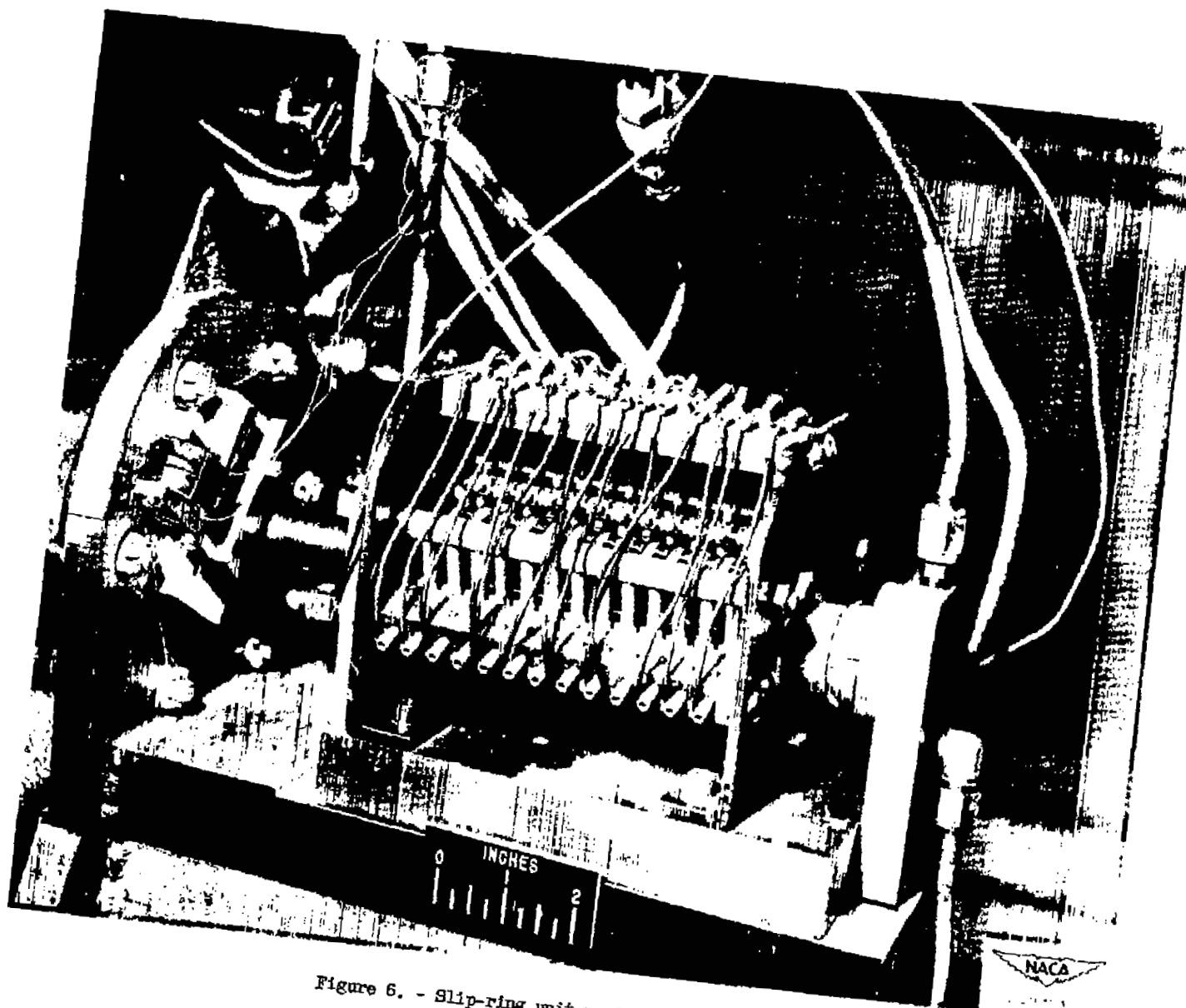
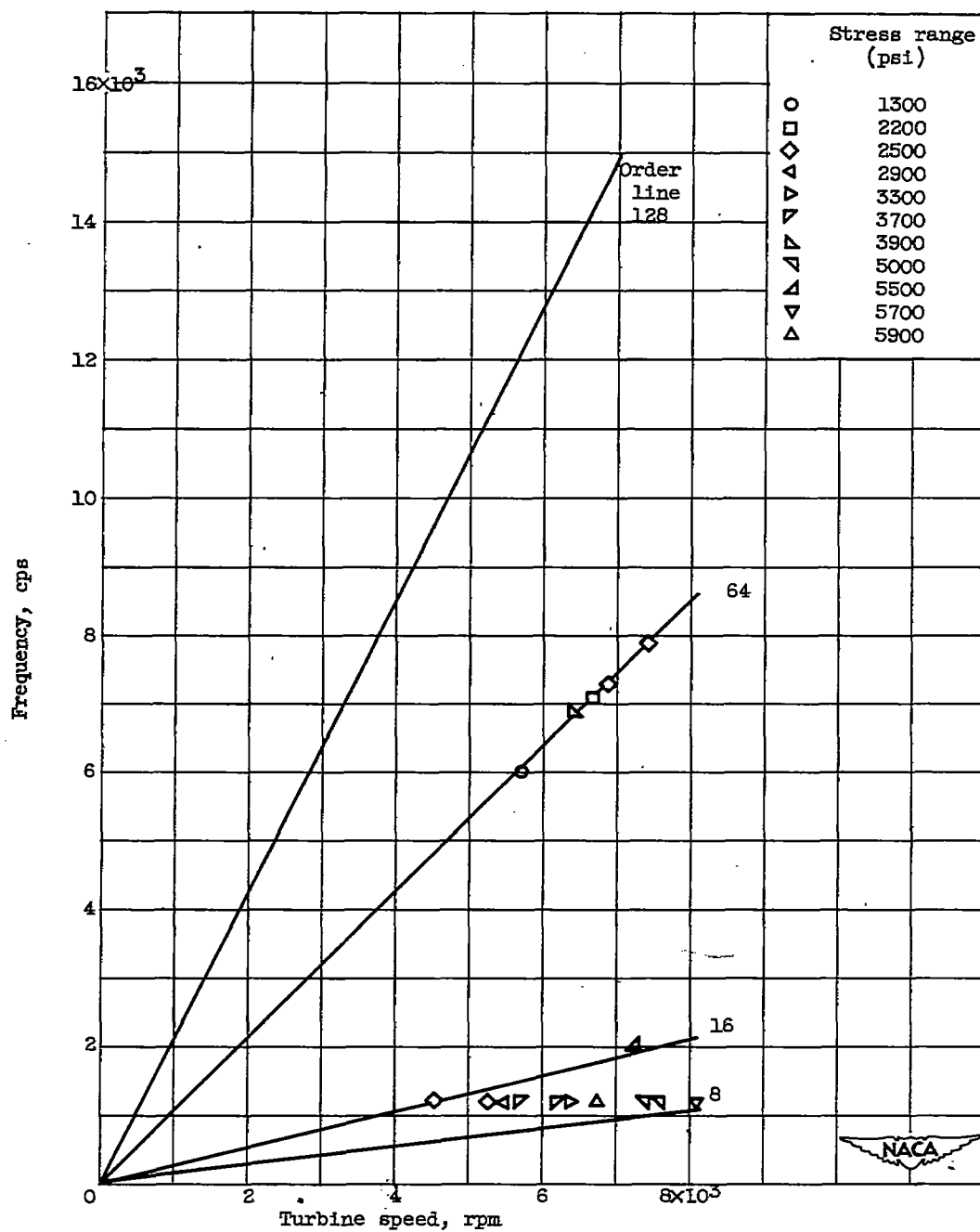
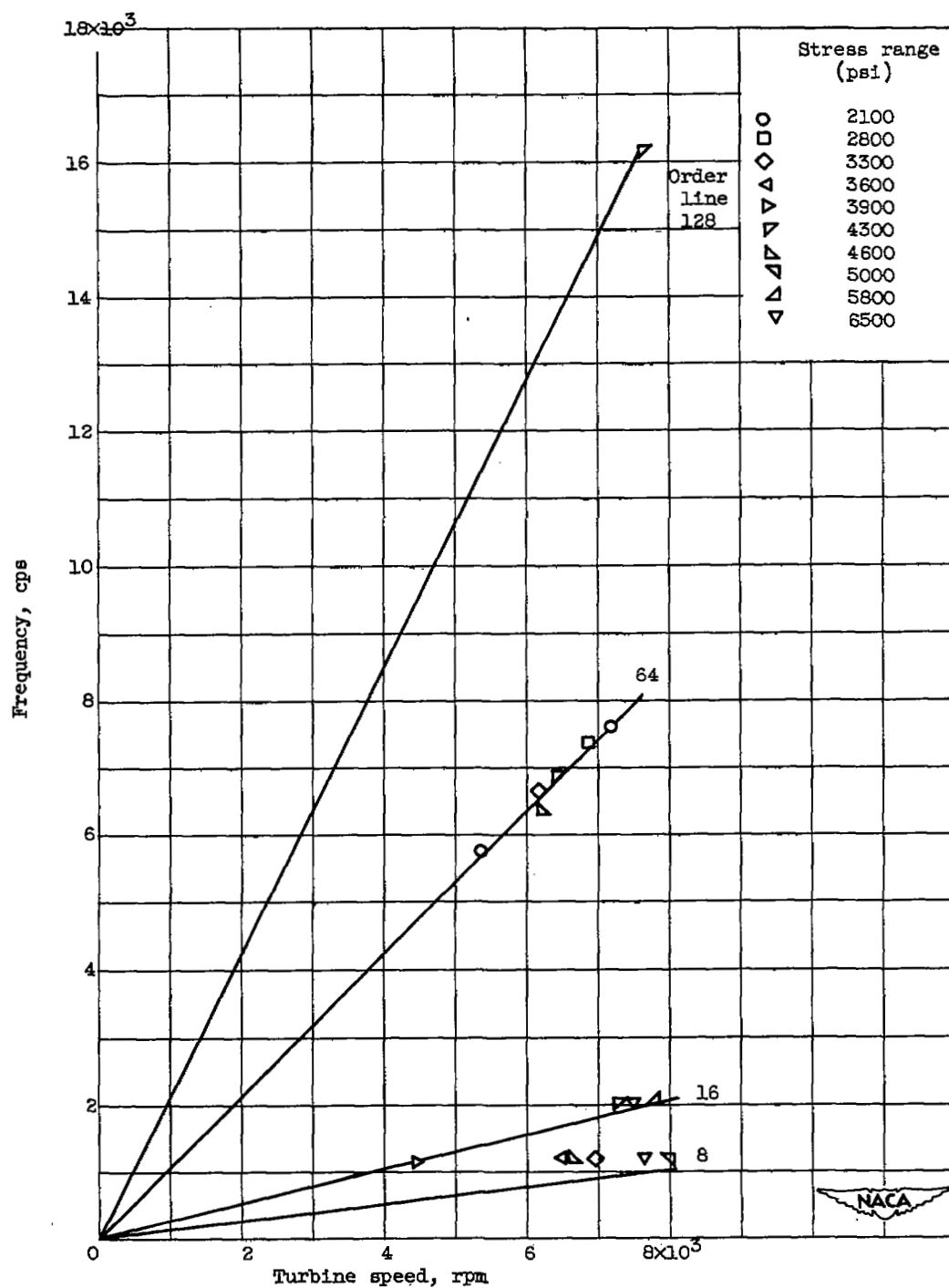


Figure 6. - Slip-ring unit used in investigation.



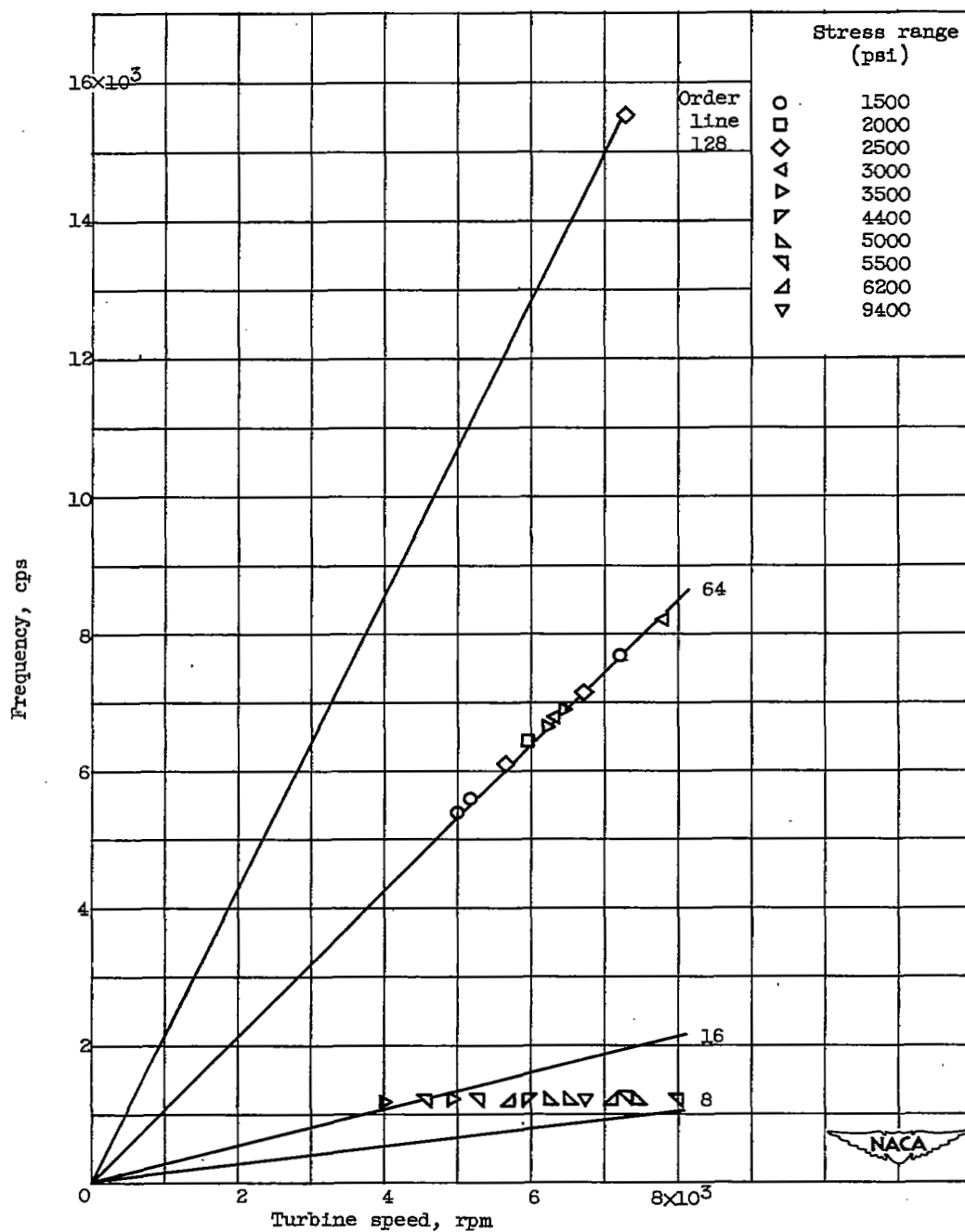
(a) Position 1 ($1\frac{1}{4}$ -in. spacing).

Figure 7. - Critical-speed diagram of vibration in design A turbine blade.



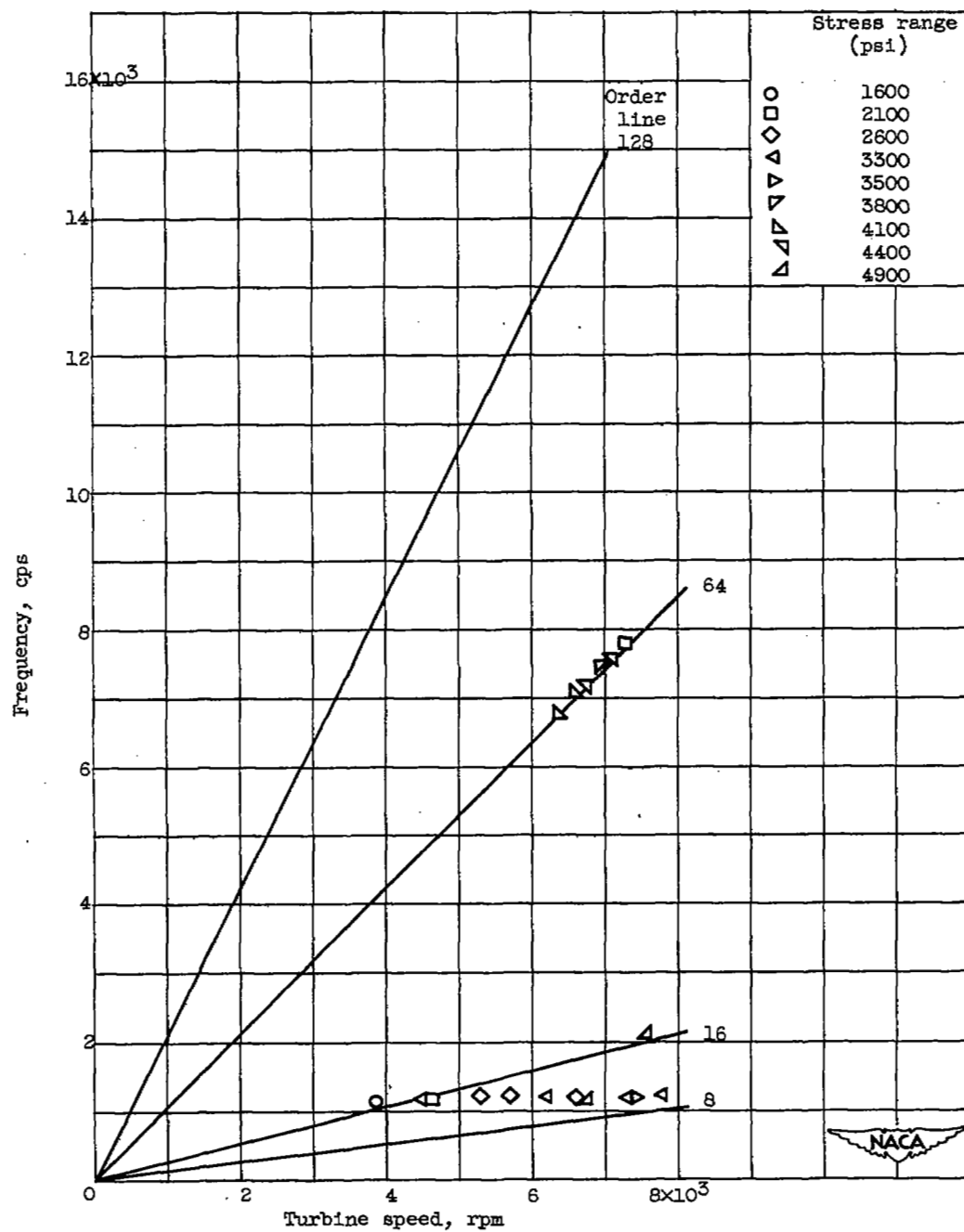
(b) Position 2 (1-in. spacing).

Figure 7. - Continued. Critical-speed diagram of vibration in design A turbine blade.



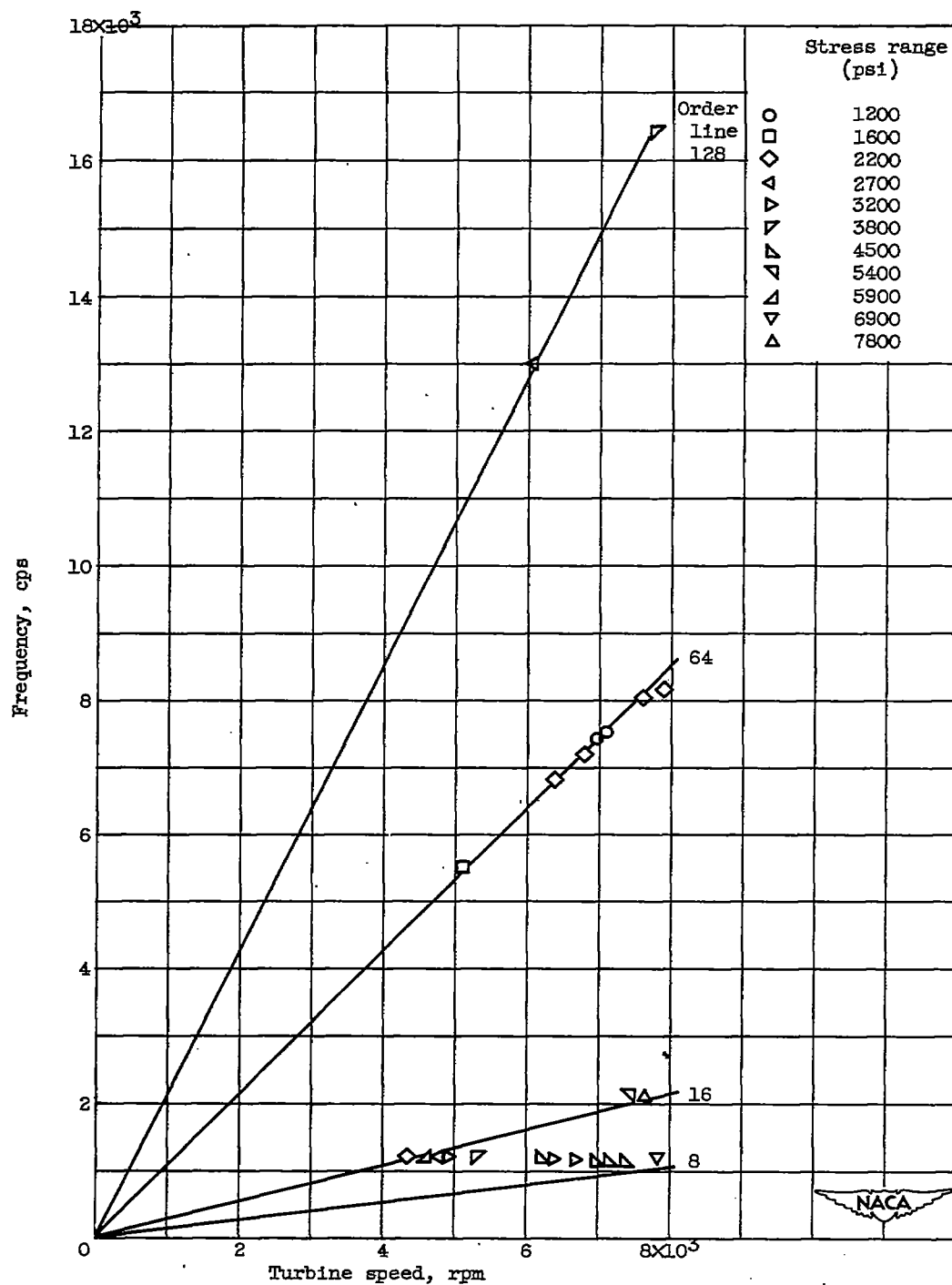
(c) Position 3 (3/4-in. spacing).

Figure 7. - Concluded. Critical-speed diagram of vibration in design A turbine blade.



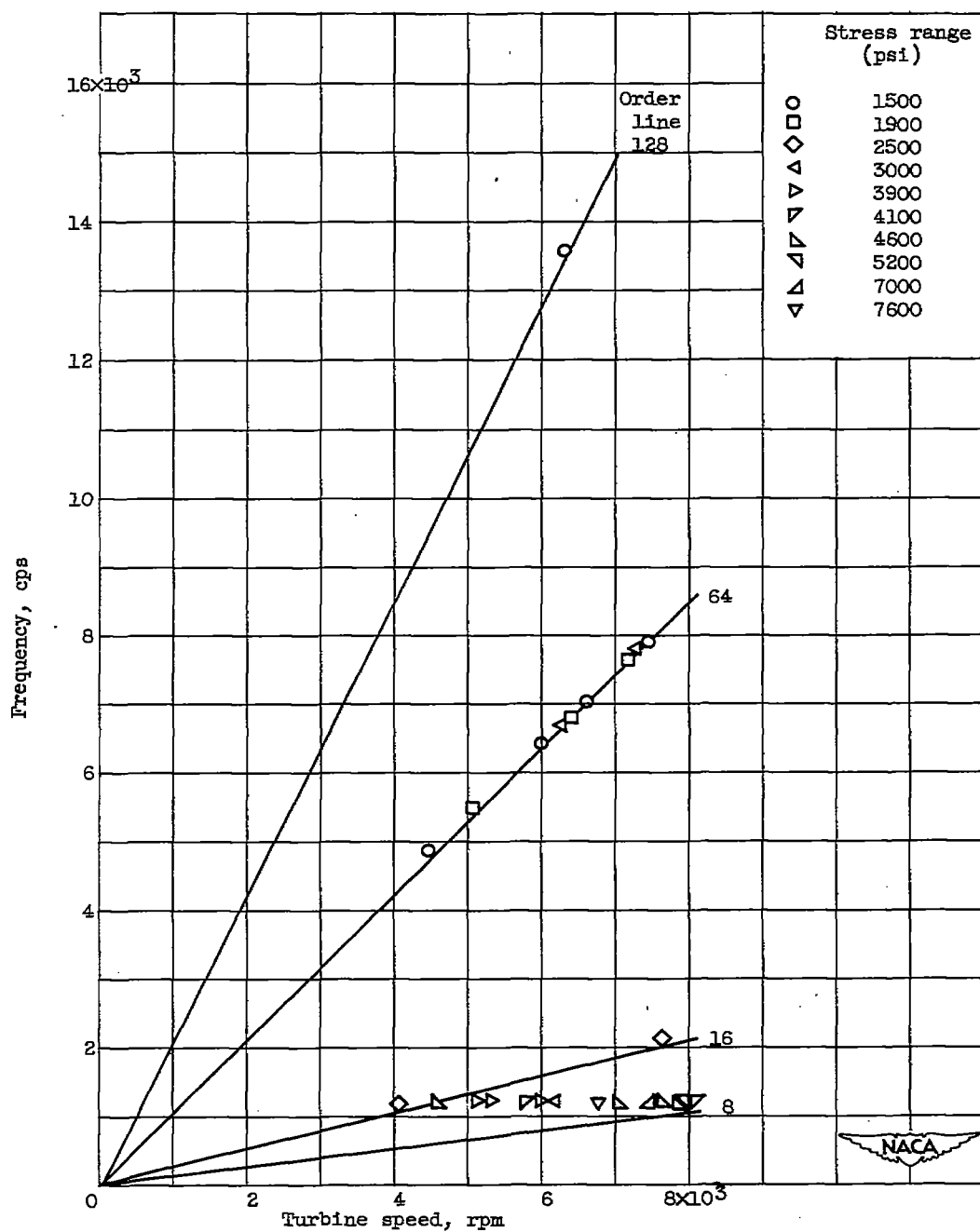
(a) Position 1 ($1\frac{1}{4}$ -in. spacing).

Figure 8. - Critical-speed diagram of vibration in design B turbine blade.



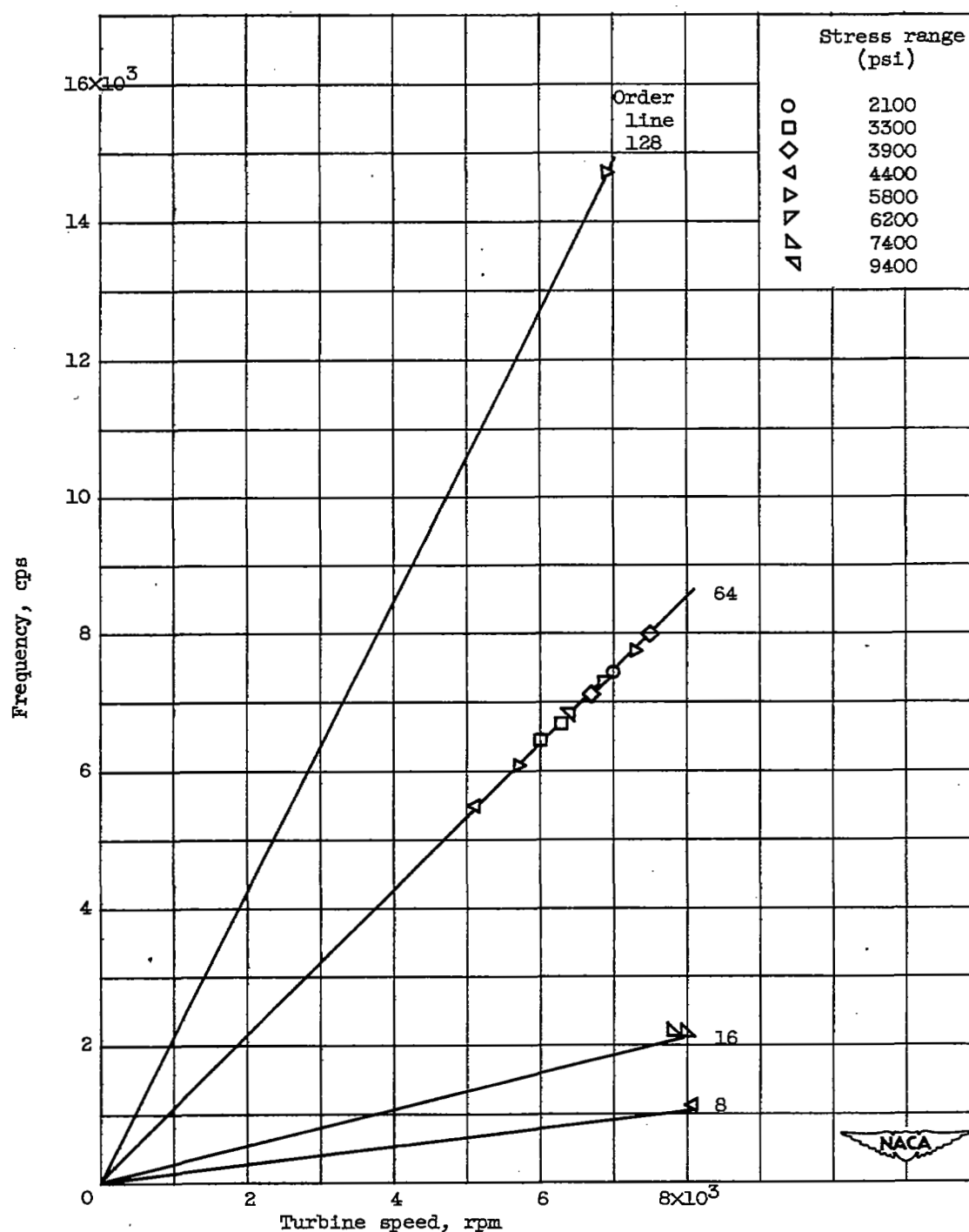
(b) Position 2 (1-in. spacing).

Figure 8. - Continued. Critical-speed diagram of vibration in design B turbine blade.



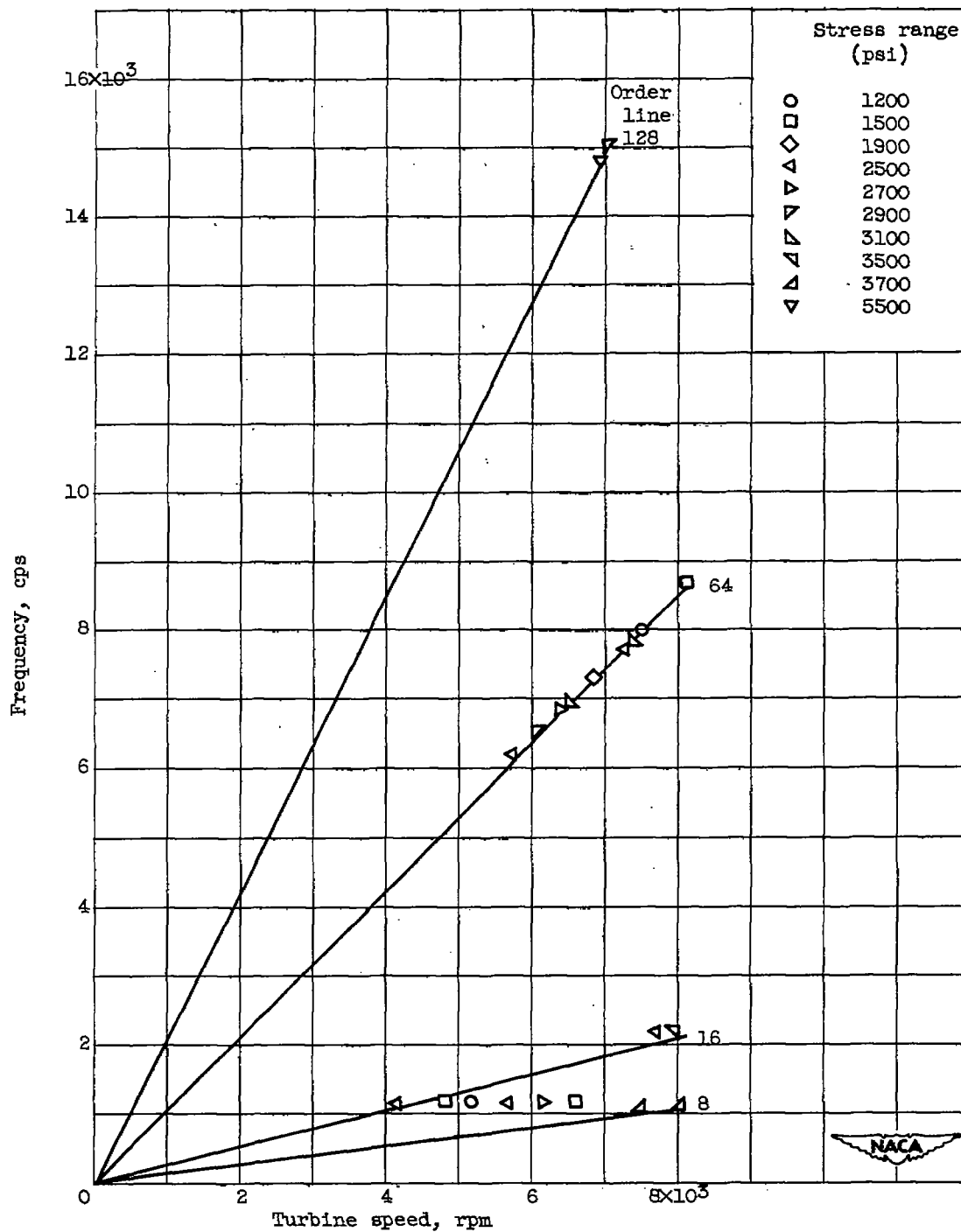
(c) Position 3 (3/4-in. spacing).

Figure 8. - Concluded. Critical-speed diagram of vibration in design B turbine blade.



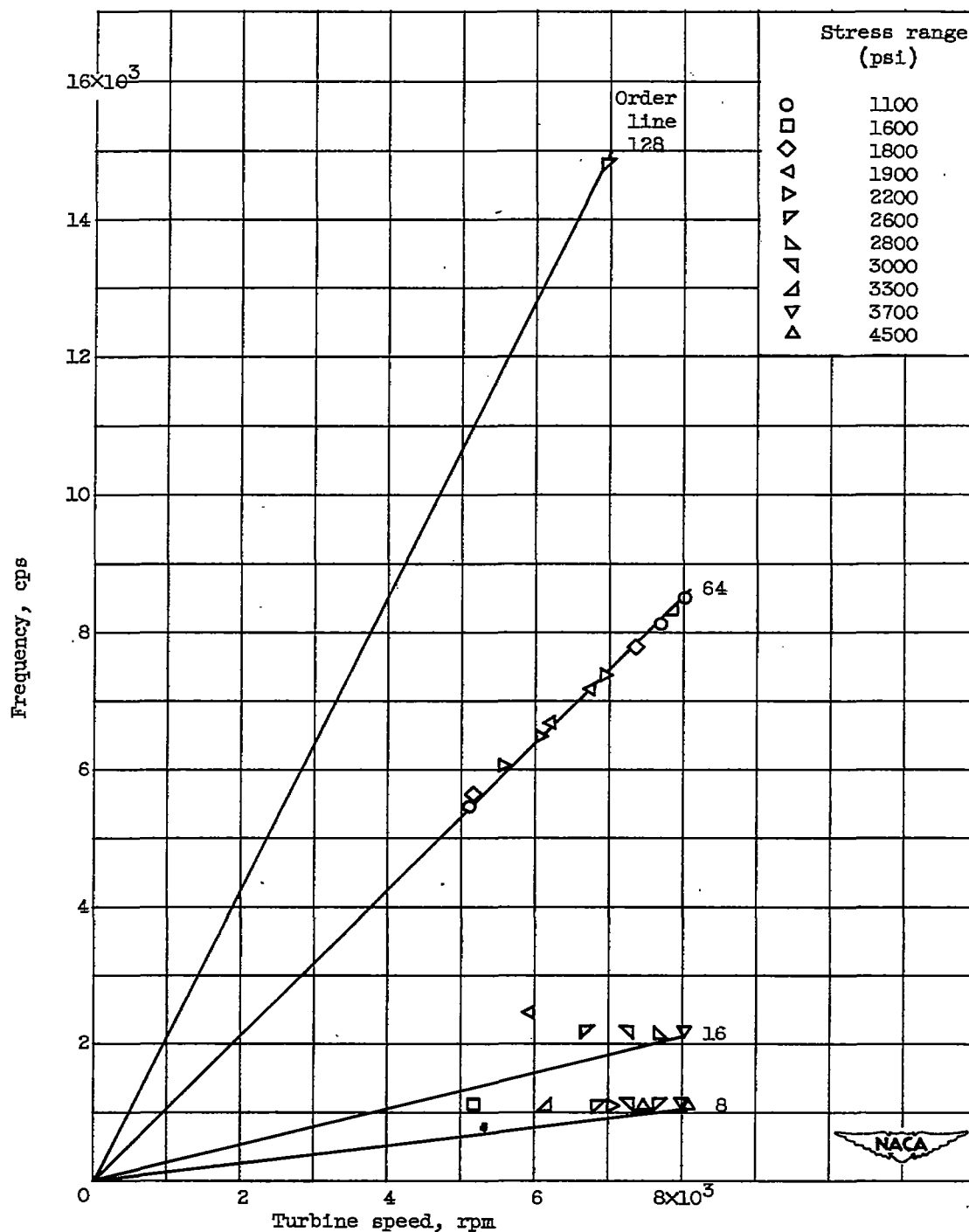
(a) Position 1 ($1\frac{1}{4}$ -in. spacing).

Figure 9. - Critical-speed diagram of vibration in design C turbine blade.



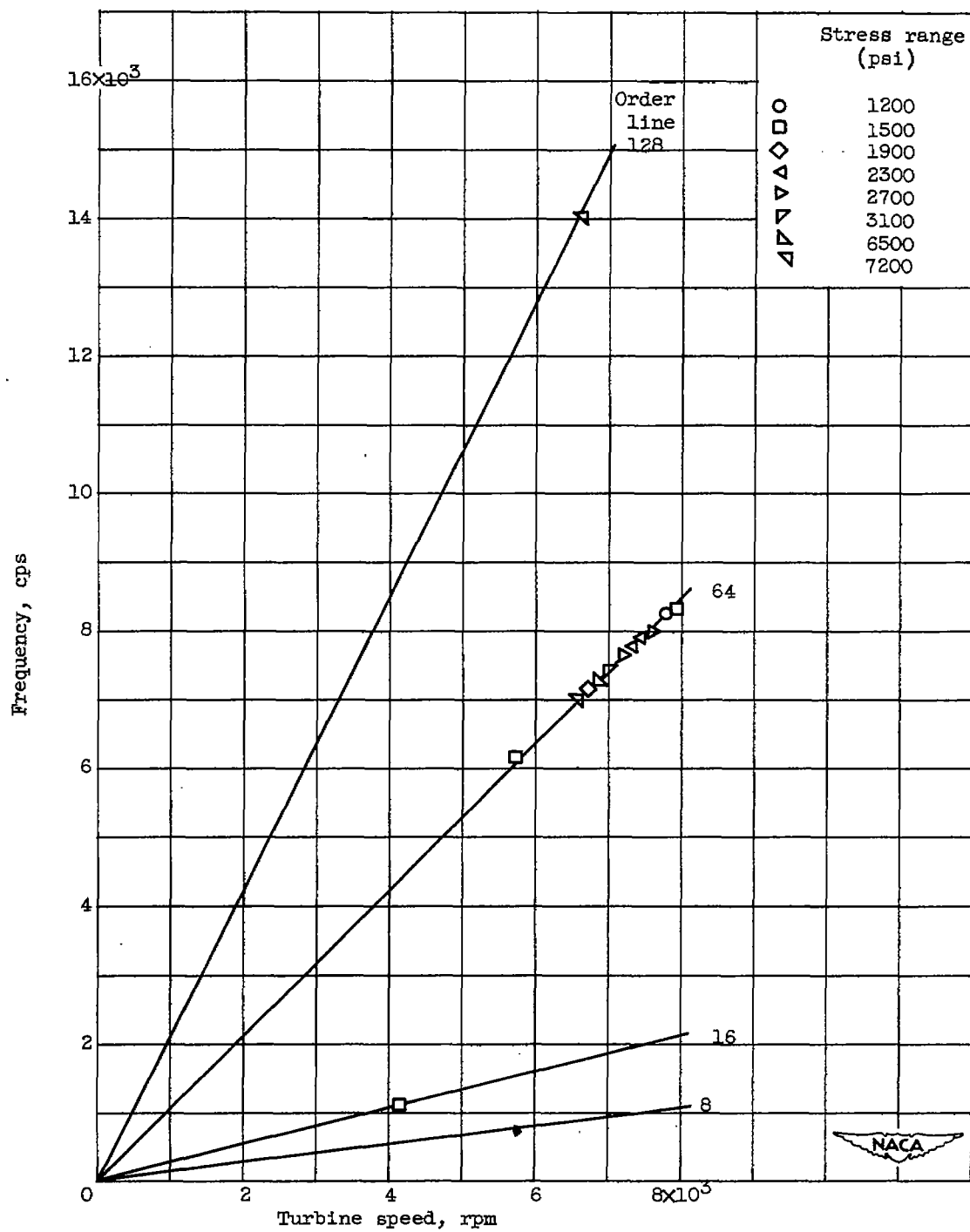
(b) Position 2 (1-in. spacing).

Figure 9. - Continued. Critical-speed diagram of vibration in design C turbine blade.



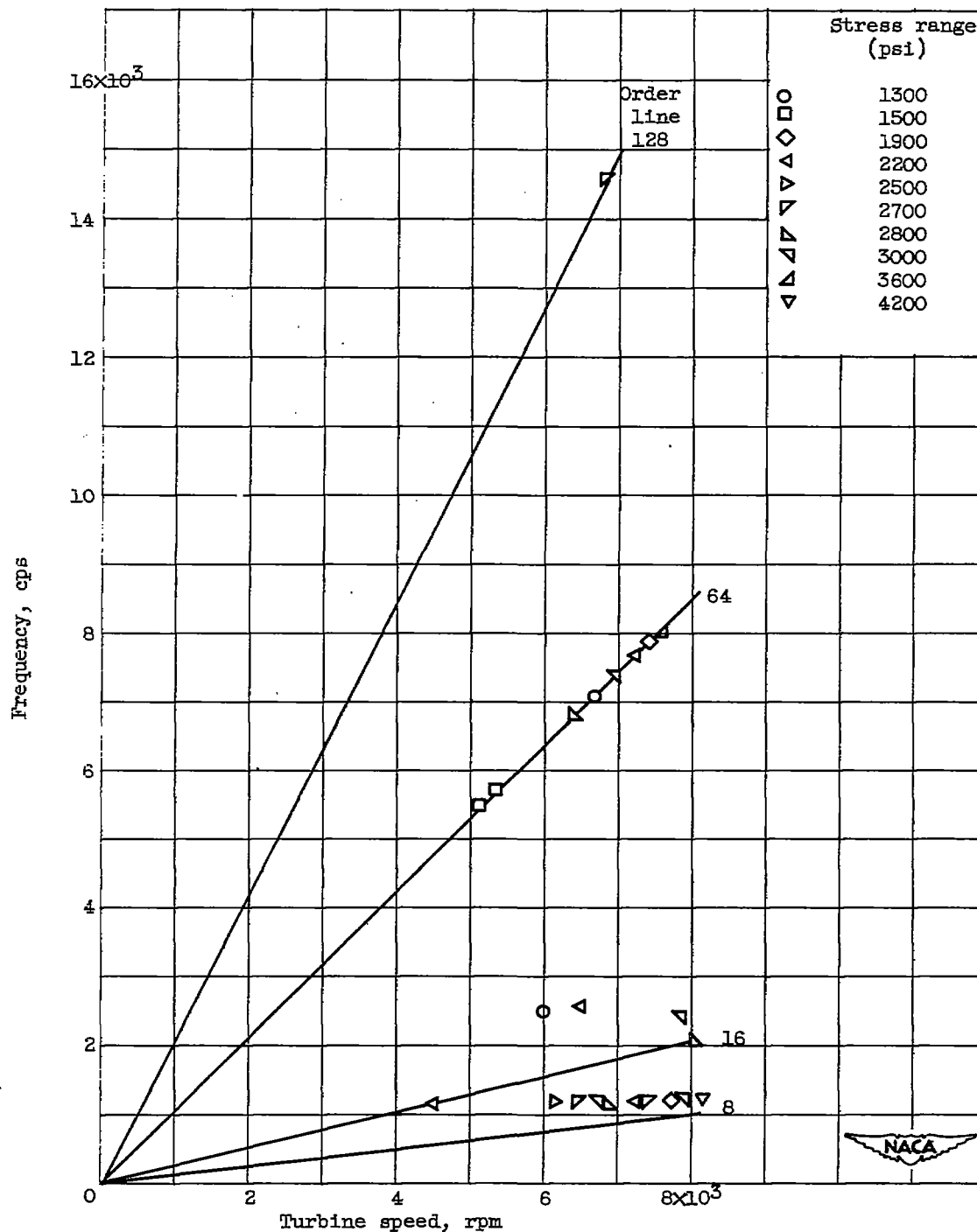
(c) Position 3 (3/4-in. spacing).

Figure 9. - Concluded. Critical-speed diagram of vibration in design C turbine blade.



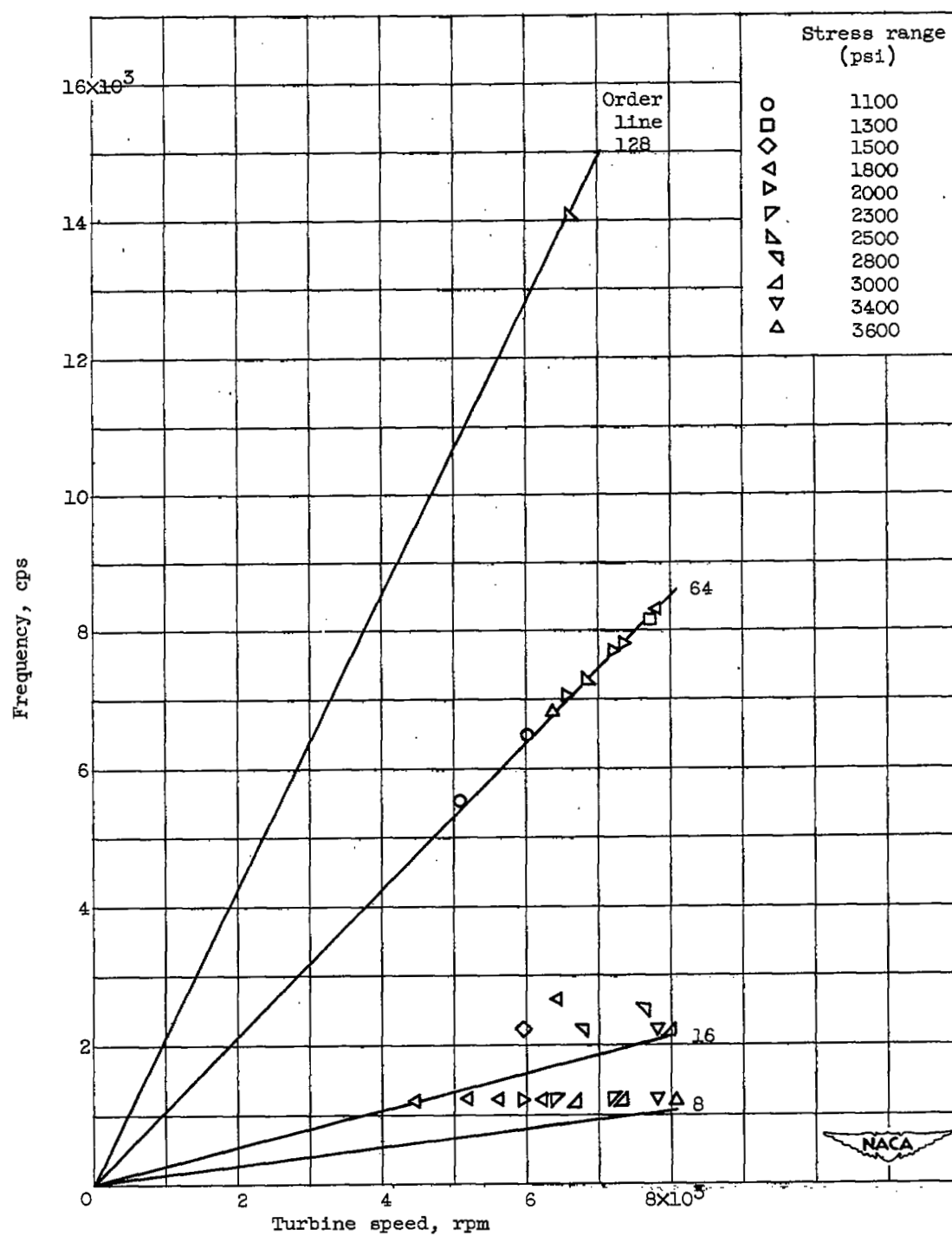
(a) Position 1 ($1\frac{1}{4}$ -in. spacing).

Figure 10. - Critical-speed diagram of vibration in design D turbine blade.



(b) Position 2 (1-in. spacing).

Figure 10. - Continued. Critical-speed diagram of vibration in design D turbine blade.



(c) Position 3 (3/4-in. spacing).

Figure 10. - Concluded. Critical-speed diagram of vibration in design D turbine blade.

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